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Horizontal Heat Exchanger Design and Analysis for Passive Containment Heat Removal Systems

Yearly Progress Report
June 1, 2003 through May 31, 2004

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Executive Summary

This project is investigating the major aspects of horizontal heat exchanger performance in passive containment heat removal from a light water reactor following a design basis accident LOCA (Loss of Coolant Accident). Since the application to passive containment heat removal is new, the following aspects of heat exchanger behavior are being investigated:

1. the condensation heat transfer characteristics when the incoming fluid contains noncondensable gases
2. the effectiveness of condensate draining in the horizontal orientation
3. the conditions that may lead to unstable condenser operation or highly degraded performance
4. multi-tube behaviors with the associated primary-side and secondary-side effects

This project consists of experimental investigations and development of mechanistic models for incorporation into industry safety codes such as TRAC and RELAP. During the first two years, two graduate students (one of which was supported by the NEER funding) and an undergraduate from Purdue University's School of Nuclear Engineering participated in the experimentation phase. Students will acquire additional experimental experience and will also receive instruction in development of physics models and use of reactor safety codes for the remainder of the project.

During the first year, an experimental facility was constructed with a single-tube test section. Initial data for local condensation heat transfer coefficients were obtained. The measured data were converted to local heat fluxes. These data demonstrated that the heat transfer rates are symmetric about the condenser tube very close to the tube inlet. Downstream, the heat transfer rates are much higher from the top side of the tube than the bottom, most likely due to additional thermal resistance from a thicker condensate film along the bottom of the tube. These data are unique and important in that they

provide multi-dimensional information (angle and distance from inlet) on condensation heat transfer rates without disturbance of the condensation phenomena by intrusive instrumentation or thermal distortion of the condenser tube geometry.

In the second year, additional measurements from the single-tube condenser test section were taken and condensate draining was further investigated to ensure that heat exchanger performance is not inhibited by water plugging. Noncondensable gases were shown to inhibit steam condensation. Calibration of the innovative thermocouples used to obtain an estimate of the condenser tube inner surface temperature proved more difficult than expected. A technique that appears to be successful is currently being employed and it is anticipated that calibrated data will be available early in the third year of the project. A six-tube test section was also constructed and preliminary data from this test facility was obtained which led to some significant observations regarding design issues. Modifications are needed to the facility to allow for refill of secondary side coolant water during operation and improved measurements of the coolant boil off rate for energy balances. The current setup is sufficient to evaluate the relative heat transfer efficiency of each tube and overall evaluation of the primary side performance.

In the third year, calibration of the single-tube test facility thermocouples will be performed and final test data will be obtained. The conditions leading to highly degraded performance or unstable condenser operation will be investigated. Modifications to the six-tube test section will be made and the tube bundle experiments will be performed. An analytical model of the heat transfer processes will be developed for implementation into reactor safety codes and concepts for a mechanistic model will be proposed. Finally, the heat transfer model will be incorporated into the TRAC or RELAP code and verified against experimental data.

This report documents the progress from June 1, 2003 through May 31, 2004.

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Acronyms

DBA	Design Basis Accident
LOCA	Loss of Coolant Accident
PCCS	Passive Containment Cooling System
SBWR	Simplified Boiling Water Reactor

Nomenclature

h	heat transfer coefficient [W/(m^2 K)]
M_a	mass fraction [-]
P	pressure [kPa]
q''	heat flux [W/m^2]
r	radial location [m]
T	temperature [K]
u	velocity [m/s]
z	axial location [m]

Greek Symbols

δ	condensate film thickness [m]
ϑ	azimuthal angle [degrees]

Subscripts

ann	annulus
wall	condenser tube wall
in	condenser tube wall inner surface
out	condenser tube wall outer surface

1. Introduction

1.1 Significance of the Project

This project is investigating the major factors of horizontal heat exchanger performance in passive containment heat removal from a light water reactor following a design basis accident (DBA) LOCA (Loss of Coolant Accident). Passive Containment Cooling Systems (PCCS) with horizontal heat exchangers are a possible contributor towards reactor safety, however there is a lack of mechanistic understanding of the heat transfer phenomena occurring in horizontal heat exchanger tubes and the need for confirmation of overall heat exchanger performance.

The significance of the research into heat exchanger performance lies in the contributions towards improved safety and reliability of essential safety functions of future LWR's. There is a strong move towards passive safety systems because the equipment is driven by failsafe forces or mechanisms such as gravity and natural circulation. Horizontal heat exchangers bring benefits over the current vertical heat exchanger designs such as lower maintenance requirements, reduced capital costs due to reduced containment building height and volume, improved security due to reduced building height and better seismic resistance characteristics. The heat exchangers investigated in this work may be used in both advanced and innovative reactors. The modeling methods that result from this study will provide a tool for evaluating system performance and will be necessary for the reactor design and licensing processes.

The importance of the condensation heat transfer investigation lies in clarification of a basic phenomenon that occurs in heat exchangers of several nuclear and non-nuclear applications.

This NEER grant is providing a significant opportunity to develop nuclear engineering students. During the first year, two graduate students (one of which was supported by the NEER funding) and two undergraduates from Purdue University's School of Nuclear

Engineering participated in the experimentation phase. An additional undergraduate student joined the project in the fall of 2003 as part of the School of Nuclear Engineering's "Undergraduate Research Experience" program. The students are acquiring additional experimental experience and receiving instruction in development of physics models and use of reactor safety codes for the remainder of the project.

1.2 Passive Containment Heat Removal

Passive systems for containment heat removal following a design basis accident LOCA are one of the strategies for achieving simplification and improving safety and reliability of future nuclear reactors. Passive systems are those that do not require any external input such as AC power sources or operator action to function. Compared with active systems, the passive designs are much simpler because they do not depend on the availability of large power supplies and they do not rely on safety-grade containment cooling systems, both of which add cost and complexity. Yadigaroglu [1999] provides a review of the various passive designs.

The driving forces for these systems are relatively small forces such as natural circulation for cooling and gravity for condensate return. In particular, the heat transfer processes are driven by small pressure and temperature differences. Thus, to achieve the needed cooling rates, heat transfer with phase change is necessary. In one of the first passive concepts, General Electric designed the Simplified Boiling Water Reactor (SBWR) PCCS with vertical heat exchangers that condense containment steam and transfer the heat to a pool outside the containment [Vierow, 1991, 1992]. This design is the basis for passive systems of several current plant designs in the US, Europe and Japan.

1.3 Condensation Heat Transfer in Horizontal Tubes

In contrast to the vertical tube situation, in horizontal heat exchanger tubes, steam condenses along the inner surface of the tube and runs along the periphery to the bottom. At the top portion of the tube, the condensate layer is thin and provides a relatively small thermal resistance. The primary heat transfer mode through the condensate layer is conduction. When the flow is stratified, the primary heat transfer modes at the bottom are conduction and forced convection; however, the thicker condensate layer at the bottom provides more heat transfer resistance, rendering the upper section of the tube a more effective heat transfer surface. The situation is further complicated by the droplet entrainment and mist formation by gas shear.

Heat transfer rates at both the top and bottom sections of the tube must be known in order to evaluate the PCCS heat transfer capabilities. This requires detailed knowledge of the local temperatures and condensate film characteristics at the condenser tube top and bottom. Ideally, data would be taken at several angles around the tube; however if the approximate location of the stratified condensate level is known, a reasonable two-region approximation for the heat transfer rates can be obtained.

An additional factor in the PCCS system must be taken into account. The intake gas mixture from the containment will contain a significant concentration of noncondensable gases. These gases accumulate along the condensate film to form a boundary layer that inhibits steam from reaching the film surface. With a lower vapor partial pressure at the condensation surface, the heat transfer rate is decreased and the heat exchanger performance is degraded.

Figure 1 illustrates a possible flow development in horizontal PCCS tubes. The thickness of the condensate layer is a function not only of the distance from the condenser tube inlet, but also of the azimuthal angle. The condensate film represents a heat flow resistance and the heat transfer rate is an inverse function of the condensate film thickness.

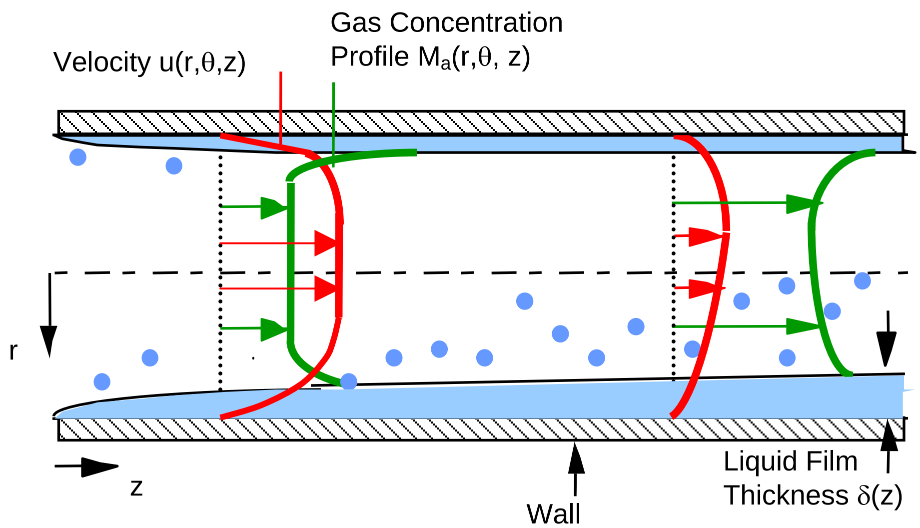

Figure 1 Profiles in Horizontal Tubes of Condensing Steam-Air Mixtures

In the PCCS situation, annular flow and stratified flow are expected to be the dominant regimes. Annular flow occurs at the test section inlet where there is minimal condensate present and the steam has a high velocity. The steam may have a high enough velocity along the condenser tube to entrain condensate droplets and form annular mist flow. Stratified flow is expected downstream. As steam condenses, the steam velocity will decrease and the condensate will run down along the sides of the tube to accumulate at the tube bottom. Stratified flow may also be observed close to the inlet for low steam flow rates.

Two main determinants of the flow regime in horizontal tubes with condensing steam are gravity and interfacial shear. Gravitational forces cause the phenomena to be asymmetric at any cross section, resulting in a stratified condensate layer at the bottom of the tube. Shear determines the flow-through of the condensate as it is pulled along by the steam as well as the surface condition of the condensate layer, i.e. the heat transfer surface and the droplet entrainment. The development of the flow regime is quite complicated, yet its effect on heat transfer rates is significant and must be understood to develop mechanistic models of the phenomena.

1.4 Advantages of Horizontal Heat Exchangers

Horizontal condensation heat exchangers have traditionally found many industrial applications, including in the process industry and the air conditioning and refrigeration industry [Kakac, 1998]. Within the nuclear industry, the steam generators of Russian-design VVER reactors are horizontal heat exchangers. The horizontal heat exchanger proposed in this project is applicable to advanced reactors and to some of the Generation IV reactors that use passive cooling. The modeling is applicable to the heat exchangers of the German SWR 1000 [Twilley, 2002].

Extensive past experience has shown that horizontal exchangers offer several advantages over vertical exchangers including less tube fouling and higher structural earthquake resistance. Heat transfer rates are expected to be very high. For the passive heat removal application investigated herein, there is an economic benefit because the shorter coolant pool allows for reduction in the containment building height and volume. Further, the reduced containment building height makes the plant less of a target for attack from the outside.

1.5 References

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2. Project Objectives and Tasks

The project's overall objectives are to: design a horizontal heat exchanger for passive containment heat removal from a light water reactor following a design basis accident LOCA; experimentally investigate the major aspects of the heat exchanger's behavior; develop analytical tools for incorporation into reactor safety codes; investigate condensation heat transfer mechanisms; and help develop nuclear engineering students and a junior faculty.

The specific tasks to accomplish these objectives are to:

a. construct an experimental facility with a single-tube test section
b. obtain fundamental data for local condensation heat transfer coefficients and condensate draining in a horizontal tube
c. construct an experimental facility with a tube-bundle test section
d. obtain data on the heat removal performance of the tube-bundle test section
e. investigate conditions leading to highly degraded performance or unstable condenser operation of the heat exchanger
f. develop a heat transfer coefficient correlation for implementation into a reactor safety code
g. incorporate the heat transfer model into a reactor safety code and verify the model against experimental data

3. Project Status

3.1 Single-tube Experiments

3.1.1 Scaling Analysis

This experimental facility was designed to simulate a single PCCS tube. The goals of the single-tube experiments are to obtain fundamental data for local condensation heat transfer coefficients and condensate draining in a horizontal tube.

The condenser tube is a straight tube and the length of the tube along which condensation can occur is 120 inches (3.04 m). Little data is available on the length necessary for complete condensation in a horizontal tube. In the 10 MW units of the vertical SBWR condensers, the 2.4 m tube length is longer than needed for long term heat removal in 2-inch diameter tubes. A length of 3 m was chosen based on calculations using the Chato correlation [1962] for condensation heat transfer in horizontal tubes with stratified flow, because PCCS steam/air flow is expected to be stratified except near the condenser tube entrance.

A 1-1/4 inch tube diameter was chosen with a 0.083 in. tube wall thickness. This wall is thick enough to install thermocouples for measuring the tube wall inner and outer surface temperatures. From these temperatures, the local heat flux through the tube wall can be accurately determined.

3.1.2 Facility Description

The major components of the test facility are: the steam supply, the noncondensable gas supply, the coolant water supply, the test section, the condensate collection system, the associated piping and water storage tanks, the instrumentation and the data acquisition system. A detailed description may be found in the Year 1 report with an overview of the test section provided below.

The test section consists of a stainless steel condenser tube and polycarbonate plastic blocks with a 2-1/2 inch hole axially through the center to form the cooling jacket.

Steam and noncondensable gas are mixed in the steam supply line and directed into the condenser tube. Steam condenses in the tube, while uncondensed steam, condensate and air flow through. The exiting condensate water is collected in a tank. Cooling water flows through the secondary-side annular cooling jacket counter current to the primary side flow.

The centerline, tube wall and coolant thermocouples are placed in sets at fourteen axial locations along the test section. These locations are closely spaced near the condenser inlet and spaced further apart with distance from the inlet. The thermocouple locations at each of 14 cross sections are shown in Figures 2 and 3.

For condenser tube inner surface temperatures, a hole is drilled nearly through the tube wall and a thermocouple with a plug design is tapped into the hole. The plug is made of stainless steel. Type T thermocouple wires (copper and constantan) are threaded through holes in the plug and soldered together to form the measurement junction. This is an innovative inhouse thermocouple design.

3.1.3 Experimental Procedures

With the steam/air mixture flowing into the test section, the condensate exits to the condensate collection tank. The condensate collection tank water level is maintained at a steady level, towards the lower end of the sight glass. Any uncondensed steam is discharged to the outside atmosphere from the test section outlet.

Control data are plotted on the LabVIEW display and monitored. When the temperatures in the steam supply and the pressure and temperature in the test section have been constant for at least 5 minutes, the system is deemed to be at steady state. Data is then recorded for each channel for at least a two-minute period.

3.1.4 Experimental Test Ranges

The tests are being performed at post-LOCA, low-pressure (up to 0.4 MPa) conditions. The target ranges of conditions are shown in Table 1. The steam flow rates correspond to the actual steam flow rates and velocities expected into a single PCCS tube.

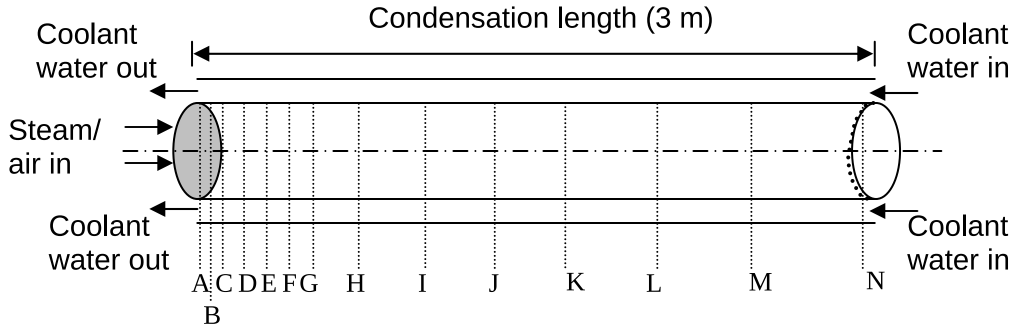

Figure 2 Axial Locations of Thermocouples in Single Tube Facility

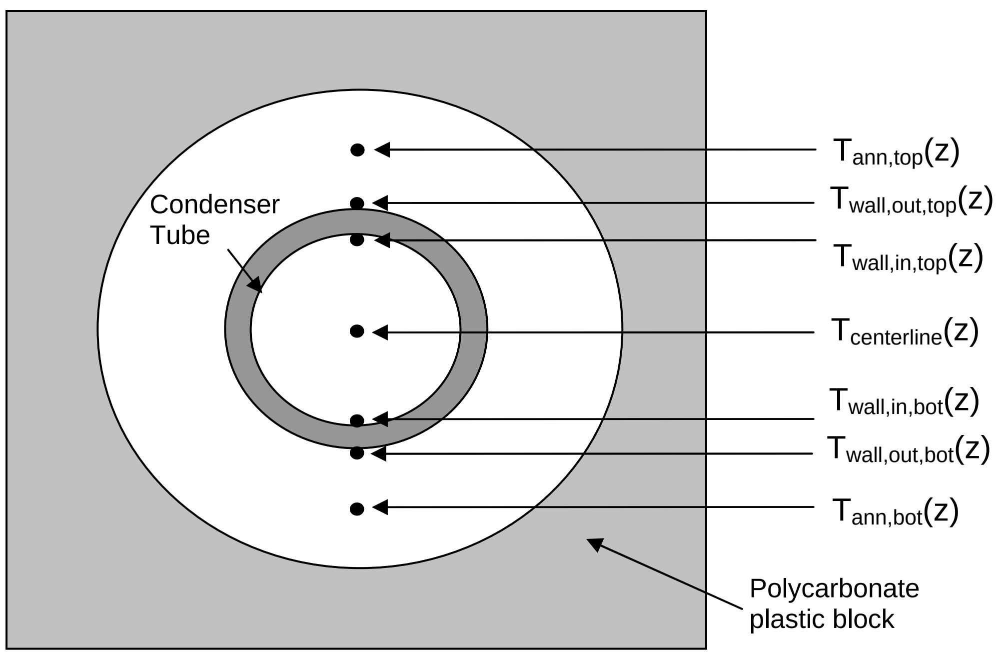

Figure 3 Cross Sectional View of Thermocouple Locations in Single Tube Facility

Table 1 Targeted Single-Tube Experimental Conditions

Parameter	Range
Primary side pressure (MPa)	0.1-0.4
Steam velocity (m/s)	1-20
Noncondensable gas inlet concentration (Vol. %)	0 – 20
Secondary side pressure (MPa)	0.2
Secondary side water flow rate (kg/s)	See note 1
Secondary side coolant inlet temperature (^oC)	10 - 20

1) Flow rate are set to provide about a 15^oC coolant temperature rise assuming complete condensation, except at the highest steam flow rates where the coolant temperature rise is about 30^oC.

3.1.5 Experimental Results

Figure 4 shows temperature profiles taken during a typical test. The temperatures confirm the expected trends. The centerline temperature is the highest temperature, although the centerline thermocouple second from the tube exit was malfunctioning. Moving from the tube centerline outward to the coolant, the temperatures decrease in order, specifically from tube centerline to tube inner wall surface to tube outer wall surface to coolant temperature.

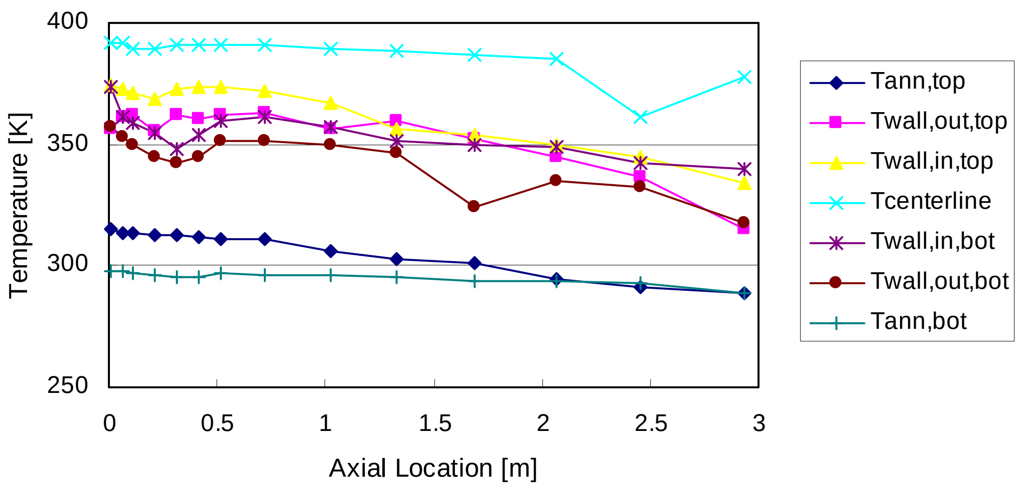

Figure 4 Temperature Profiles along the Test Section in Single Tube Tests
0.2 MPa, 11.05 g/s, 5.8% inlet air volumetric concentration

The second important trend is that the temperatures at the top of the tube are higher than the temperatures at the bottom of the tube in the stratified region. Near the inlet, the flow is annular and nearly symmetric around the tube. Thus, top and bottom tube temperatures are similar. Downstream, the flow is stratified and the condensate thermal resistance reduces the heat transfer through the lower portion of the tube. Thus, the lower section exhibits lower temperatures.

The data was reduced as discussed in the Year 1 report and local heat fluxes and local heat transfer coefficients are plotted in Figures 5 and 6 respectively. The heat flux for phase change heat transfer is high and Figure 5 shows that the heat flux is on the order of magnitude typical of condensation in heat exchangers. The difference between the flux at the top and bottom of the tube are readily apparent.

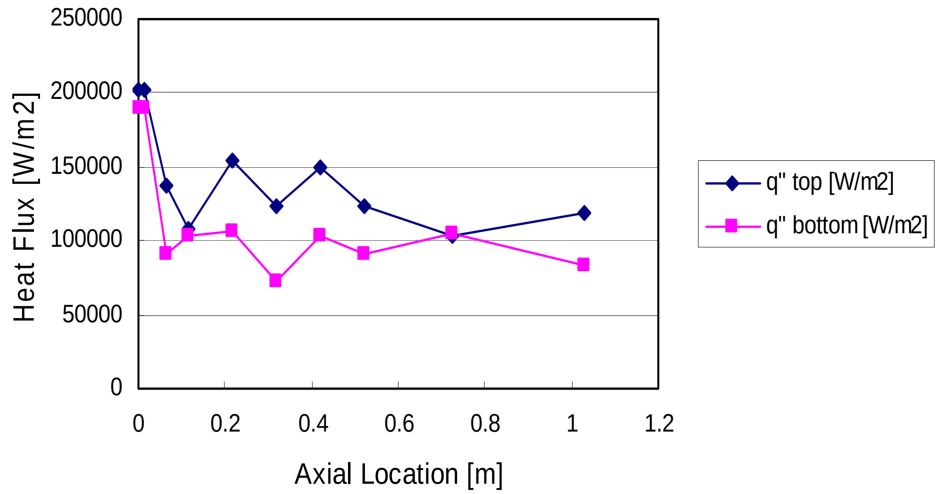

Figure 5 Local Heat Flux Profiles in Single Tube Tests
0.2 MPa, 11.05 g/s, 5.8% inlet air volumetric concentration

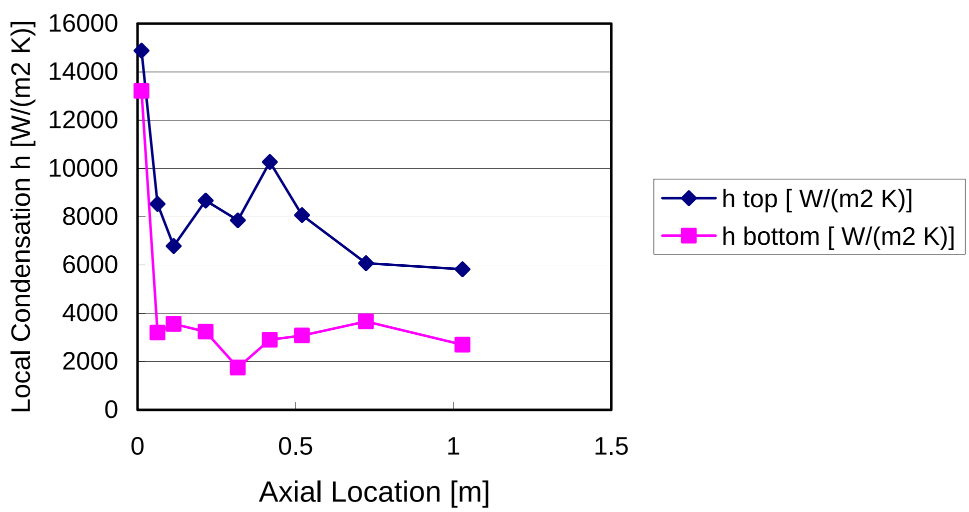

Figure 6 Local Heat Transfer Coefficients in Single Tube Tests
0.2 MPa, 11.05 g/s, 5.8% inlet air volumetric concentration

Figure 6 shows that the local heat transfer coefficients are also highly dependent on the radial location. With a thinner film along the upper surface, the condensation process is much more efficient than along the bottom of the tube.

3.1.6 Calibration of Thermocouples

The curves were expected to be smoother than in Figures 5 and 6 and occasions of outer surface temperatures exceeding inner surface temperatures such as in Figure 4 are not physically possible. Calibration of the tube inner surface thermocouples, which were designed and manufactured inhouse, should correct these discrepancies.

Several attempts to carry out the calibration failed because symmetry could not be obtained for the tube cross section in a horizontal position. The condenser tube has been repositioned vertically and a new secondary side for calibration has been designed.

A calibration device (Figure 7) has been manufactured and preliminary calibration has been done on a short condenser tube to confirm the feasibility of the device. The condenser tube is positioned vertically and a short coolant annulus is attached to the condenser tube. The annulus is filled with water. During the calibration, steam flows through the condenser tube and the coolant in the annulus is heated to saturation temperature. While in a boiling mode, the heat transfer along the short section may be assumed to be uniform. The steam generated from the coolant annulus is condensed in a secondary condenser and collected in a graduated cylinder. The collection process is timed with a stop watch so that the evaporation rate can be calculated. Then the average heat flux can be calculated. A linear relationship between the calculated heat flux and the measured temperature difference should be found. The heat flux can be changed by changing the steam pressure and velocity in the condenser tube. By moving the short coolant section along the condenser tube, calibration curves for each set of thermocouples can be obtained. These calibration curves can be used to correct the experimental data.

The experiment on the short section has been finished and preparations are underway for calibration of the full test section.

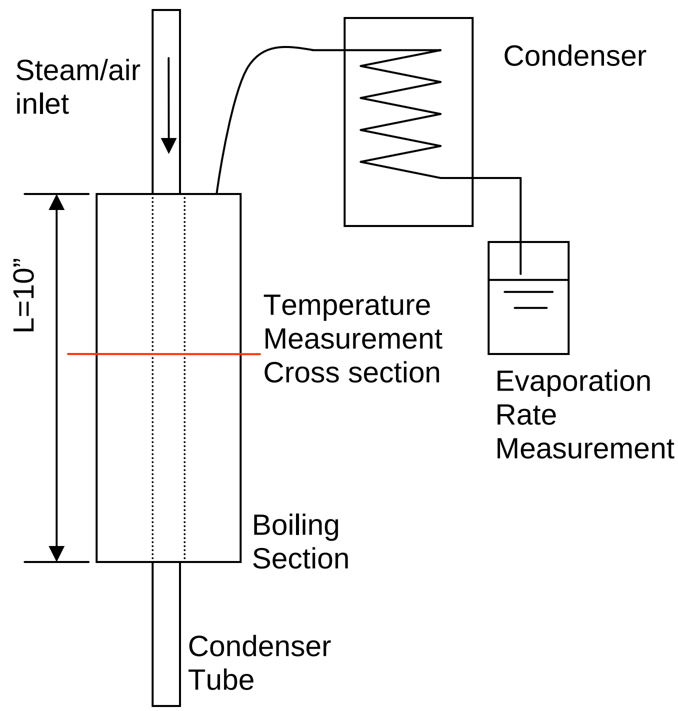

Figure 7 Schematic of Calibration Device for Single Tube Facility

3.2 Tube Bundle Experiments

3.2.1 Scaling Analysis

This experimental facility was designed to simulate a portion of a PCCS tube bundle. The condenser tube is a straight tube and the active heat transfer length of the tube is 3.96 m (156 inches). The tube length was decided based on the results of the single-tube experiments. Using the length needed in the single-tube experiments for complete condensation under various pressures and noncondensable gas concentrations, calculations were performed to estimate an optimal length and diameter.

A 1-1/2 inch tube diameter was chosen with a 0.12 in. tube wall thickness. The diameter was selected because scoping calculations show that this diameter will provide a steam/gas mixture velocity between 1 and 12 m/s under LOCA conditions and this should be sufficient to promote condensate draining at the exit but not too high to prevent nearly complete condensation. A 2-inch diameter has been proposed for many vertical heat exchangers, however concern arose about the draining characteristics if complete condensation were to occur within a short distance in the tubes.

3.2.2 Facility Description

3.2.2.1 Integral Test Loop

As shown in Figure 8, the major components of the test facility are: the test section, steam supply, noncondensable gas supply, the coolant water supply, the condensate collection system, the associated piping and water storage tanks, the instrumentation and the data acquisition system. All heated components are thermally insulated with Microlok fiberglass insulation. Each of the components and the instrumentation are described in the following sections.

3.2.2.2 Test Section

The test section consists of six stainless steel condenser tubes and a secondary-side pool in which coolant water removes heat by boiling heat transfer. Tube specifications are given in Table 2.

Table 2 Tube Bundle Test Section Specifications

Condenser tube outer diameter	38.1 mm (1.5 in.)
Condenser tube wall thickness	3.05 mm (0.12 in.)
Condenser tube heat transfer length	3.96 m (156 in.)
Number of condenser tubes	6
Pool length	3.96 m (156 in.)
Pool width	0.13 m (5.25 in.)
Pool height	0.30-0.41 m (12-16 in.)

Steam condenses in the tubes, while uncondensed steam, condensate and air flow through. The exiting condensate water is collected in a separate condensate collection tank for each tube.

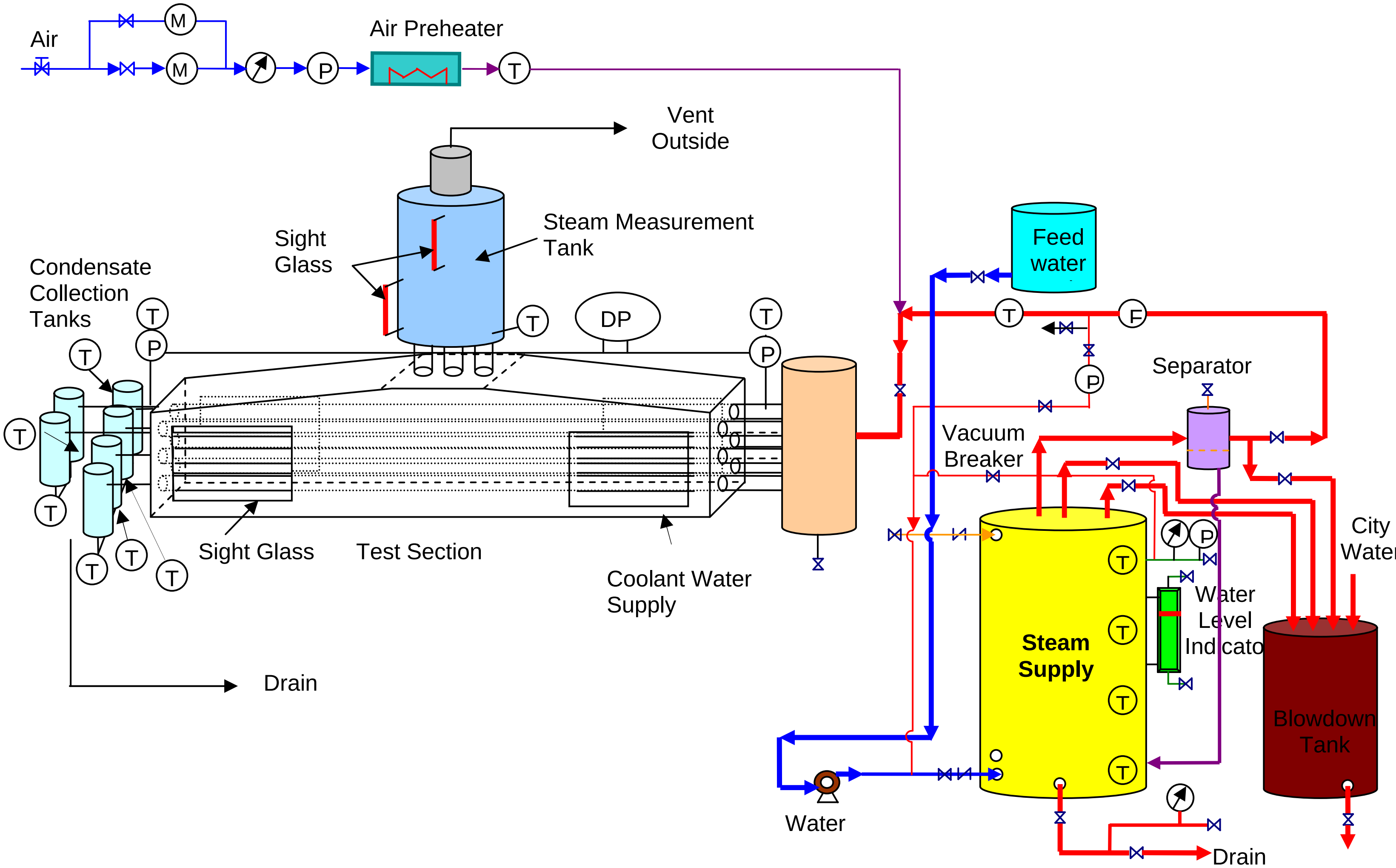

Figure 8 Facility Layout for Tube Bundle Tests

Heat is removed by a coolant pool that surrounds the six condenser tubes. The pool is heated to saturation eventually boils. The cover of the secondary side pool is inclined upward towards the center. This allows for venting of steam from the pool through the vent above the test section. Four sight glass windows were installed in the pool structure to allow for visual observation of the secondary side boiling and void distributions around the condenser tubes. Two sight glasses are located at each end of the pool structure, on opposite sides. One of these sight glasses is shown in Figure 9, along with the condenser tubes inside.

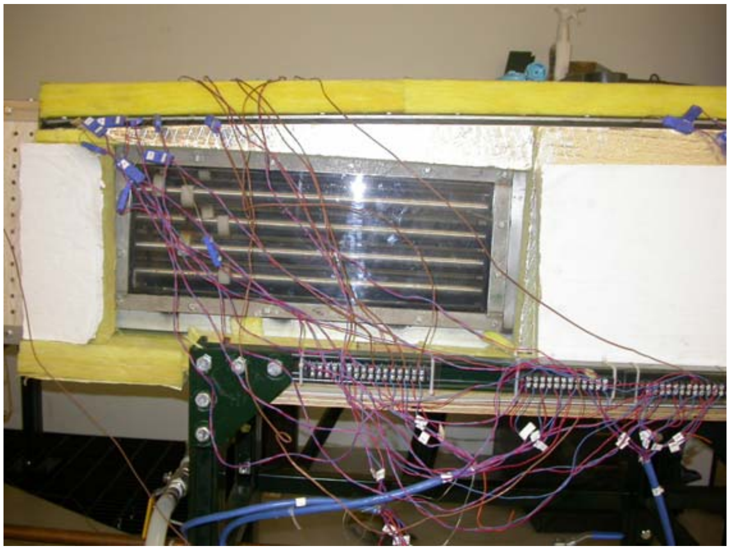

Figure 9 Sight Glass on Secondary Side Pool Structure

Along the active heat transfer sections of each of the tubes, 5 equidistant thermocouples were installed to measure bulk centerline temperature profiles. In contrast to the single-tube experiments, secondary side boundary condition is pool boiling after the initial startup period. The tube outer surface temperature at the tube top and the coolant temperature are measured at three axial locations. These correspond to the inlet, outlet and middle cross sections. Following the basic ideas of Kuhn and Peterson's method [1995, 1997] for the tube outer wall temperatures, a groove is made on the condenser tube outer surface and a thermocouple is anchored in the groove.

All temperatures are measured with Type T (copper-constantan), 1.0 mm, ungrounded sheathed thermocouples. Figure 10 shows the locations of the thermocouples at a cross section where there are tube outer surface and coolant thermocouples.

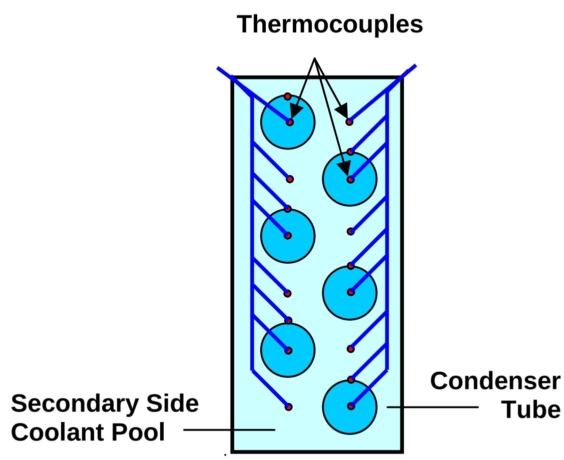

Figure 10 Cross Sectional View of Thermocouple Locations for Tube Bundle Facility

A rubber silicon gasket between the pool and the pool cover prevents leakage of pool water and steam. Extensive leak testing was performed to achieve a leak-free test section.

3.2.2.3 Steam Supply

The steam supply consists of a pressure vessel, immersion heaters and a control panel. The pressure vessel was designed by the contractor and manufactured by Kennedy Tank and Manufacturing Co., Inc.

The steam supply before insulating is shown in Figure 11. The pressure vessel shell is Schedule 10 stainless steel 304 pipe, 60 inches in height and 24 inches in diameter, along with two 24-inch, Schedule 10 stainless steel end caps that were welded to the top and bottom of the body. The vessel was built for 150 psi and hydrotested at 180 psi for 12 hours.

Three 8-inch, flanged immersion heaters manufactured by Watlow Process Systems were purchased for the current project. The heaters have inconel sheaths. Each heater has a total output of 50 kW. Two of the heaters have two 25 kW circuits and one heater has eight 6.25 kW circuits. The power level is adjustable in 6.25 kW increments up to 150 kW.

Figure 11 Steam Supply Vessel (before insulating)

Noncondensable gas is injected into the steam supply line and directed into an inlet plenum. The inlet plenum is designed to provide a uniform distribution of steam/noncondensable gas into the six tubes by use of two spargers that force mixing in the inlet plenum and avoid preferential jetting of steam/air into any of the tubes.

3.2.2.4 Noncondensable Gas Supply

The noncondensable gas is air. The University's air compressor is rated at 135 psia and a pressure regulator prevents a surge of high pressure air from entering the steam line. Two mass controllers were purchased to enable good regulation of air inflow over the entire test matrix. A 1.2 kW finned tube Chromalox heater was installed in a horizontal pipe section downstream of the mass controllers to heat the air to the incoming steam temperature.

3.2.2.5 Coolant Water Supply

De-ionized water is used for the secondary side of the test section.

3.2.2.6 Condensate Collection System

As steam condenses in the tubes, the condensate flows out each tube into its respective condensate collection tank, driven by the momentum of condensate water, any uncondensed steam and the noncondensable gas. The tanks are made of Schedule 10, 6-inch SS304 pipes and are vertically mounted as shown in Figure 12. The water levels in the tanks are controlled by valve positions on the drain lines and each tank has a sight glass with a measuring tape for measurement of the rate of change of the water level.

As the test proceeds, condensate flows out the test section and collects in the condensate collection tanks. By watching the water levels and controlling the valve positions on the drain line, the operators can maintain steady condensate water levels in the condensate collection tanks. As steady state data is being recorded, condensate water is not drained and the change in condensate level with time is recorded. From this, the condensate flow rate may be calculated. Measurements are taken over several-minute time durations, ensuring accurate measurement. The changes in the water levels do not affect the steady state because venting maintains a constant pressure.

Figure 12 Condensate Collection Tanks for Tube Bundle Facility

3.2.2.7 Instrumentation

A total of 82 temperatures, two flow rates, three absolute pressures and one differential pressure are recorded. The data is recorded by a data acquisition system assembled from National Instruments components. The LABVIEW software is used to display, scan and save data.

Flow rates are recorded for the inlet steam and the inlet noncondensable gas. For the inlet steam flow rate, a 1-in. (25.4 mm) Foxboro vortex flow meter, model 83F-A, is used.

Two absolute pressure measurements are taken on the steam line. A Honeywell absolute pressure transducer is installed near the vortex flow meter and an OMEGA absolute pressure transducer, model PX303 with a range of 0-680 kPa, is installed just upstream of the steam inlet to the condenser tube. For pressure differential across the test section, a Honeywell differential pressure transmitter, model STD924, with a range of 0 to 400 in. (0 to 0.0996 MPa) H_2O is used. Although the expected pressure drop falls in only in the lower 1/20 of this range, technical sales representatives state that accuracy standards are maintained at the low end of the instrument's pressure range. A Dwyer pressure transducer is placed on the air line.

3.2.3 Experimental Procedures

3.2.3.1 Pretest Procedures

In preparing the equipment for a run, the steam supply pressure vessel is filled with water to a water level high enough to keep the heaters covered throughout the testing. The condensate collection tank, condensate drain lines, and pressure transducer lines are all primed with water. Power supplies are turned on and all data acquisition channels are checked for proper functioning.

Heaters are turned on to a predetermined power level to heat the water in the pressure vessel. Once atmospheric pressure is surpassed in the steam supply by a few kPa, the valve to the blow-down tank is opened and steam is discharged to the blow-down vessel long enough to vent at least 10 times the volume of the steam supply gas space. This process purges the steam supply of air and ensures a pure steam delivery. The steam supply is pressurized with steam until the desired pressure is achieved.

As the steam supply is pressurizing, air flow is started and the air pre-heater is turned on to bring the air flow into the test section to the proper temperature. The secondary side pool is filled with de-ionized water. Once the steam supply pressure is at a prescribed level, the steam line is opened to the test section to start the approach to steady state.

3.2.3.2 Steady Operation Procedures

With the steam/air mixture flowing into the test section, the condensate exits to the condensate collection tank. Any uncondensed steam is discharged to the outside atmosphere from the test section outlet.

Only minor adjustments need to be made to the valve on the steam line, the condensate drain valve positions and the test section steam/gas outlet valve. The air temperature is checked periodically to ensure that it is at the steam temperature.

Key to obtaining a steady state is the position of a regulating vent valve at the test section outlet. The position must be set so that the flow rate out balances the air flow rate into the test section. Otherwise, the test section depressurizes if the valve is open too much and pressurizes if the valve is open too little.

Control data are plotted on the LabVIEW display and monitored. When the temperatures in the steam supply and the pressure and temperature in the test section have been constant for at least 5 minutes, the system is deemed to be at steady state. Data is then recorded for each channel for at least a two-minute period.

Regarding measurement of the condensate collection rate, while steady state data is being recorded, the condensate collection tank drain valve is closed and the change in the water level is recorded at regular intervals. The steady state was found to be insensitive to the water level change. Measurement of the drainage rates are taken over a 2 to 5-minute time duration, ensuring accurate measurement. The six condensate collection tanks are assigned to three students so that each person is recording the water level change for two tanks.

Consecutive tests can be run by changing the steam mass flow rate and/or the air mass flow rate.

3.2.3.3 Shutdown Procedures

System shutdown commences by closing the steam line valve to isolate the steam supply from the test section. The steam supply power is then shut off.

The air pre-heater is turned off and once the pre-heater has cooled to below 40^{o}C, the air flow is terminated. The state of the steam supply is monitored until it is in a safe enough state to stop monitoring.

After the steam supply has depressurized to atmospheric pressure, the vacuum breaker valve opens. The vacuum breaker lets air into the steam supply as it cools down and pressure decreases below atmospheric pressure, to prevent a large vacuum in the steam supply tank.

3.2.4 Experimental Test Ranges

The tests are being performed at post-LOCA, low-pressure (up to 0.4 MPa) conditions. The target ranges of conditions are shown in Table 3. The steam flow rates correspond to the actual steam flow rates and velocities expected into a single PCCS tube.

Table 3 Targeted Tube Bundle Experimental Conditions

Parameter	Range
Primary side pressure (MPa)	0.1-0.4
Steam flow rate (g/s)	6 - 65
Steam velocity (m/s)	1-13
Noncondensable gas inlet mass fraction	0.0-0.10
Secondary side pressure (MPa)	0.1

3.2.5 Experimental Results

Figure 13 shows centerline temperature profiles taken during a preliminary test. The conditions were a primary side pressure of 200 kPa, steam flow rate of 12 g/s and 5% air mass fraction. The data show that the centerline temperatures are similar for all six tubes and verify that the test section design can provide local data for heat exchanger performance evaluation. The outlet temperatures are below saturation temperature at 200 kPa (120.2^{o}C), indicating that much of the steam condensed. Tube 2 appears to have steam flowing into it with a superheat of 3-6^{o}C. This is larger than the expected thermocouple error and the cause is being investigated.

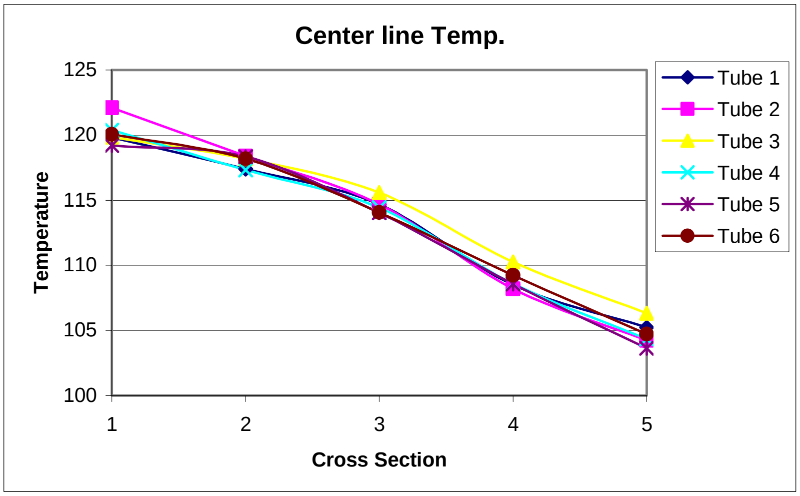

Figure 13 Centerline Temperature Profiles in Six Condenser Tubes
0.2 MPa, 12 g/s, 0.05 air mass fraction

Since some problems were found, such as a hidden loop seal that prevented draining of two of the condenser tubes, this test is not discussed further.

A second test was performed with the same conditions as the first test, but a steam flow rate of 24 g/s. The centerline profiles of Figure 14 are similar and they indicate that essentially all of the steam will be also condensed under this higher flow rate. The condenser tube outer surface temperatures in Figure 15 require additional evaluation. Decreasing profiles along the tubes were expected. In Figure 16, the pool temperatures show that the pool is at saturation and the thermocouple readings are all consistent to within experimental error.

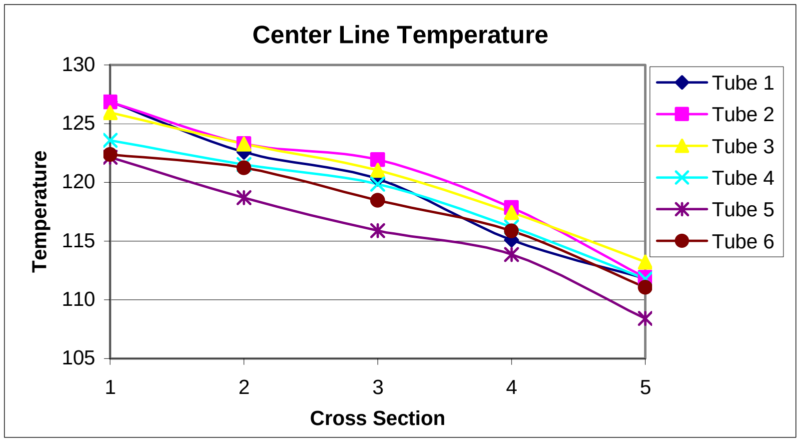

Figure 14 Centerline Temperature Profiles in Six Condenser Tubes
0.2 MPa, 24 g/s, 0.05 air mass fraction

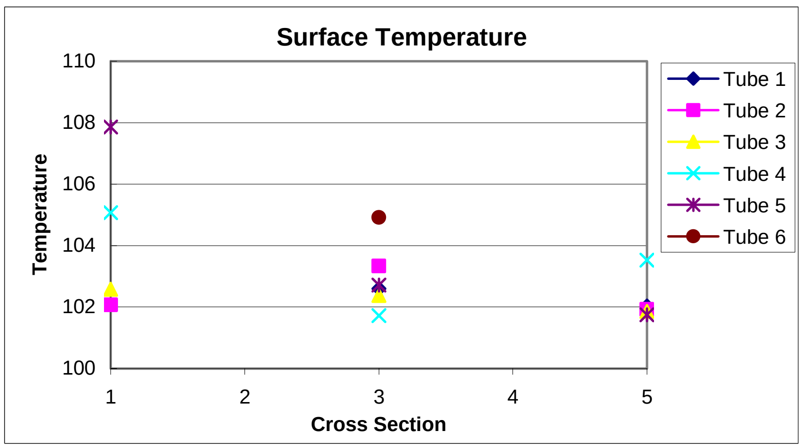

Figure 15 Tube Surface Temperature Profiles in Six Condenser Tubes
0.2 MPa, 24 g/s, 0.05 air mass fraction

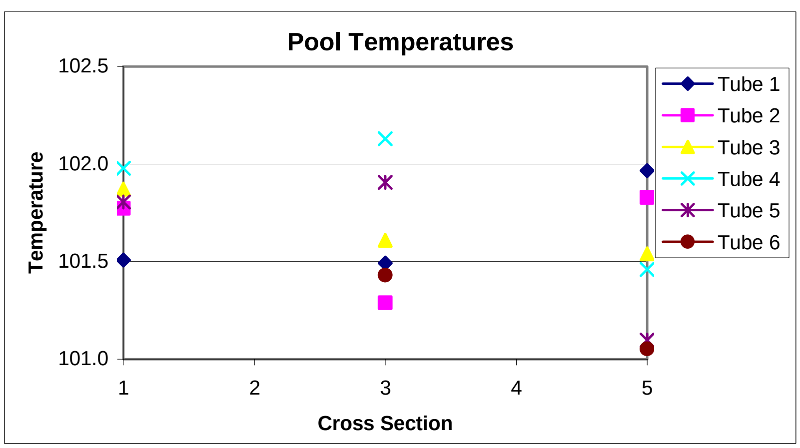

Figure 16 Pool Temperatures in Tube Bundle Experiments
0.2 MPa, 24 g/s, 5% air volumetric concentration

Figure 17 shows the condensate collection rate for each condensate collection tank. From this data, the total rate of condensation may be calculated. Equations are included in each figure for the collection rates in units of [cm/s]. System energy balances may be performed by comparing this energy removal rate with the evaporation rate of pool water. Due to design problems, the secondary side evaporation rate can not be measured and the facility is being modified to render measurement possible.

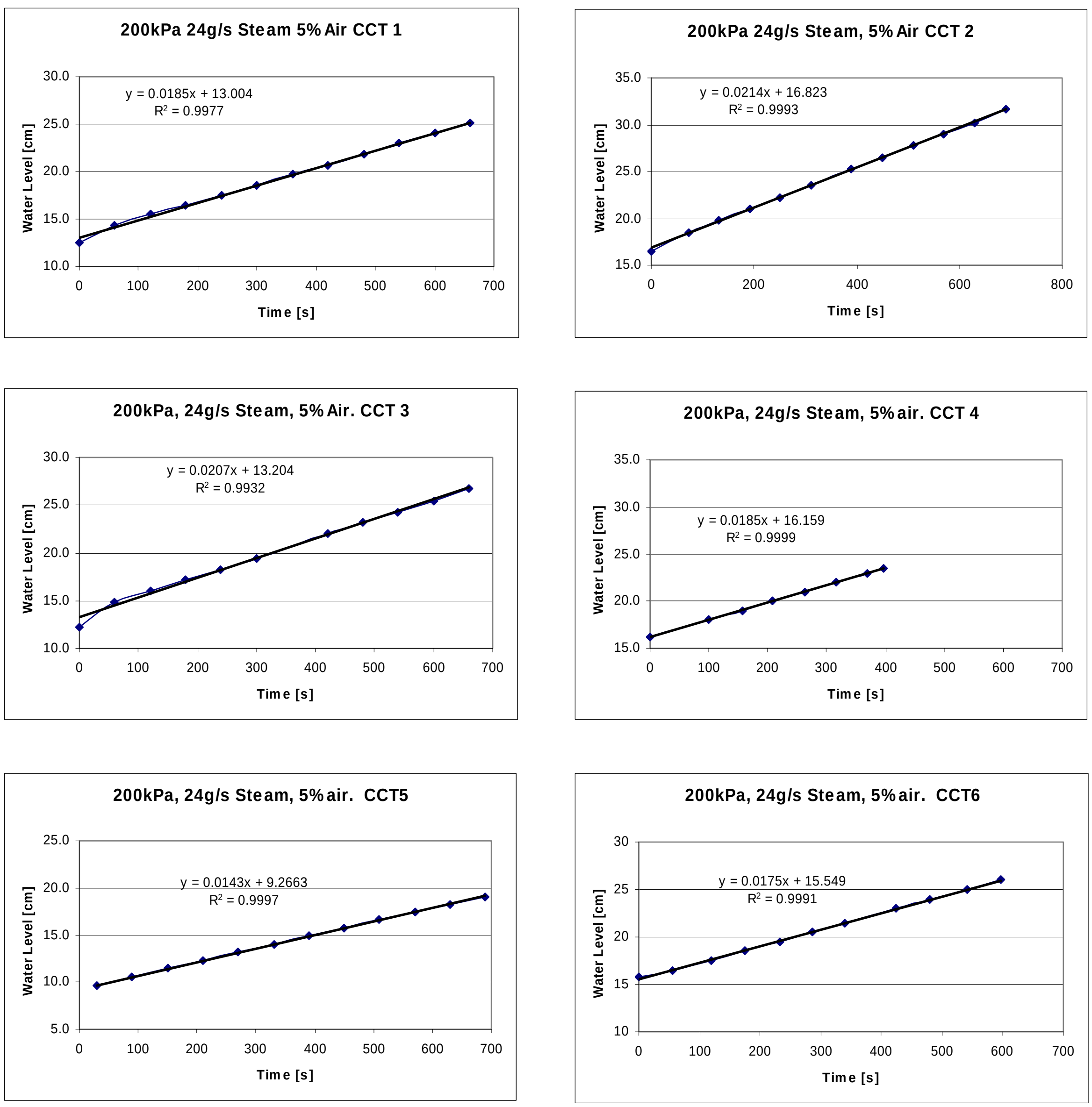

Figure 17 Collection Rate in Six Condensate Collection Tanks

In this test, the steam flow rate from the steam supply was 24.11g/s and the rate of condensate collection was 23.57g/s. It is evident that most of the steam was condensed. Comparing the collection rates with the previous test, the rate in each tube approximately doubled, in proportion to the increased inlet steam flow rate.

3.3 Issues for Future Single-Tube and Tube-Bundle Testing

For the single-tube experiments, calibration of the condenser tube inner surface thermocouples must be completed so that final data can be obtained and analysis models developed. This is delaying progress on the modeling front.

For the tube-bundle experiments, improvements must be made to the facility to allow for more complete data. The value of the tube surface temperature data must be reconsidered. For operation at higher steam flow rates, both primary and secondary side refill procedures must be developed. While the data has been shown to be valuable in a qualitative sense, quantitative evaluation methods must be developed.

3.4 References

Chato, J. C., *Jnl. of American Society of Heating, Refrigeration and Airconditioning Engineers*, p. 52, Feb., 1962.

Kuhn, S. Z., V. E. Schrock, P. F. Peterson, “An Investigation of Condensation from Steam-Gas Mixtures Flowing Downward Inside a Vertical Tube”, Nuclear Engineering and Design, Vol. 177, pp. 53 - 69, 1997.

Kuhn, S. Z., “Investigation of Heat Transfer from Condensing Steam-Gas Mixtures and Turbulent Films Flowing Downward Inside a Vertical Tube”, Ph.D. thesis, Univ. of CA, Berkeley, 1995.

4. Milestone Status

In the second year, the tube-bundle test section was constructed. Shakedown testing was performed and initial tests were conducted. This produced preliminary data from which to begin developing analysis methods.

Table 4 shows a complete list of project milestones, anticipated completion dates and actual completion dates.

Table 4 Project Milestones

ID Number	Task / Milestone Description	Planned Completion	Actual Completion	Comments
1	**Experimental Studies**			
1.1	Single-tube Experiments			
1.1.1	Construction of Facility	12/31/02	05/01/03	Steam supply delay and late arrival of grad student
1.1.2	Shakedown Testing	03/31/03	05/15/03	
1.1.3	Data Reduction Software Dev.	06/31/03	05/31/03	
1.1.4	Heat Transfer Coef. Expt.	12/31/03		Preliminary tests completed by 05/31/03. Calibration efforts ongoing.
1.1.5	Stable Operation Experiments	03/31/04		
1.2	Tube Bundle Experiments			
1.2.1	Construction of Facility	06/30/04	05/31/04	
1.2.2	Shakedown Testing	09/30/04	05/31/04	
1.2.3	Tube Bundle Experiments	03/31/05		
2	**Analytical Model Development**			
2.1	Heat Transfer Coef. Correlation	03/31/04		Awaiting final test results.
2.2	Mechanistic Model Development	05/31/05		
2.3	Model Verification and Analysis	05/31/05		
3	**Reports**			
3.1	Yearly Progress Report	08/29/03	08/12/03	
3.2	Yearly Progress Report	08/29/04	08/20/04	
3.3	Final Report	08/29/05		

5. Future Work

During the third year, calibration of the condenser tube inner surface thermocouples and measurements of condensation heat transfer coefficients in the single-tube test facility will be completed. Condensate draining will be investigated to ensure that heat exchanger performance is not inhibited by water plugging. The conditions leading to highly degraded performance or unstable condenser operation will be investigated in the "Stable Operation Experiments" using the single-tube facility.

The tube bundle test facility will be modified based on the results of the preliminary testing. Testing under steady state conditions will be performed. Conditions leading to highly degraded performance or unstable condenser operation of the heat exchanger will be investigated.

An analytical model of the heat transfer processes will be developed for implementation into reactor safety codes. Finally, the heat transfer model will be incorporated into the TRAC or RELAP code and verified against data from the Tube Bundle Experiments.

6. Patents

None.

7. Budget Data

7.1 Funding Spent to Date

Phase/Budget Period			Approved Spending Plan			Actual Spent to Date		
			DOE Amount	Cost Share	Total	DOE Amount	Cost Share	Total
	From	To						
Year 1	June 1, 2002	May 31, 2003	90,064	0.00	90,064	65,590	0.00	65,590*
Year 2	June 1, 2003	May 31, 2004	85,296	0.00	175,360	58,751	0.00	58,751*
Year 3								
		Totals				124, 341	0.00	124,341

*Expenditures are lower than budgeted because no students were on the project for several months of Years 1 and 2. This situation will not occur in Year 3.

7.2 Spending Plan for the Next Year

Month	Estimated Spending
June 2004	16,500 (includes 1 mo. faculty summer salary)
July 2004	6,500
August 2004	6,500
September 2004	21,500 (includes major equipment purchase)
October 2004	21,500 (includes major equipment purchase)
November 2004	6,500
December 2004	6,500
January 2005	6,500
February 2005	6,500
March 2005	6,500
April 2005	6,500
May 2005	6,500